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The overall goal of research - development of the basic principles of thermal processing of concrete products using air heated in the collector solar energy [1]. Research the features two varieties of this technology.

The first kind of technology - thermal treatment of concrete products is carried out using air heated in the collector of solar energy, and if necessary, use air-heater (electroheater). In the absence of incoming solar energy collector to the air-heater is the main source of heat. For methods accelerating the solidification of concrete products are necessary to determine modes their thermal processing that enable the shortest time to recruit concrete selling compressive strength. In the studies must take into account the presence of heat emission by hydration cement.

The second kind of technology - thermal treatment of concrete products is carried out using air heated in the collector solar energy and heat released during cement hydration. In the absence of incoming solar energy collector to heat treatment of concrete products takes place only using the heat released during cement hydration. For this method, accelerating the solidification of concrete products must:

- To determine the minimum and maximum possible date set concrete stripping and handling compressive strength;
- The conditions under which it is advisable to use this method.

Construction of equipment for heat treatment of concrete products using air heated in the collector solar energy should be based on the analysis of heat exchange processes that occur in them.

The use of solar energy for accelerating solidification of concrete products allows saving money in the process.

PECULIARITIES HEAT EXCHANGE IN SOLAR COLLECTOR AND IN CHAMBERS FOR HEAT TREATMENT CONCRETE PRODUCTS HEATED AIR

Chapter 1. Literature review

One way to use solar energy in the production of concrete and concrete products in tropical areas is the heating of these products in light-transparent insulating coating ([2] - [6], etc.). Light-transparent coating can be made of silica glass, polymer films, fiberglass, plexiglass.

To accelerate the solidification concrete and concrete products used as combined helio-thermal treatment of the products with the use of an intermediate carrier ([2], [3], [7] - [9], etc.).

In the patent [10] proposed to make heat treatment hydro-insulated concrete and concrete products using water heated in the collector solar energy. If necessary, apply an additional source of heat - water heater. In the patent [1] proposed to make heat treatment hydro-insulated concrete and concrete products using air heated in the collector solar energy. If necessary, is applied an additional source of heat - air-heater. Results of laboratory studies in this direction are given particular in the sources [11] - [13].

In the book [14] analyzed in detail the processes of heat transfer in a plane solar energy collector type "letter - pipe" and are given heat balances for other collectors. This balance - heat balance flat solar energy collector, which heats the air. The specified thermal balance is made under the condition the regime of collectors - stationary. However, the temperature regime of flat solar energy collector, which heats air to heat treatment of concrete products - transient that must be considered.

In the analysis of heat transfer processes in the chamber for accelerating solidification of concrete products should be considered at the selection of heat during hydration of cement ([15] - [26], etc.).

Chapter 2. Definitions intensity heating the air in a plane solar energy collector

Determination intensity heating the air in a plane solar energy collector with the following heat balance: light-transparent cover; heat-accepting metal plate and the insulation layer; overall heat balance of solar energy collector.

2.1. Scheme of collector solar energy

Consider a plane collector solar energy, which heats air for further use in thermal treatment of concrete products (Fig. 2.1). The air moves in space between a single layer of light-transparent coating and heat-accepting layer - metal plate. When a metal plate is a layer of insulation.

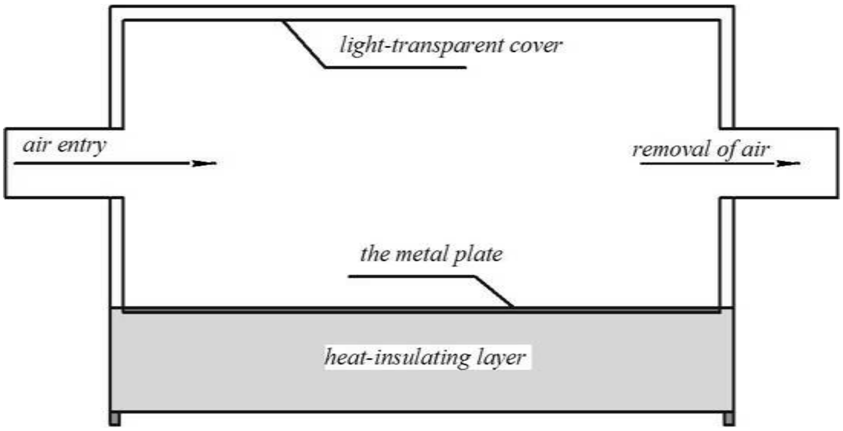


Fig. 2.1. Scheme of collector solar energy

Heated in the collector of solar energy air through the fan is directed to the camera, which is designed for accelerating solidification concrete products. In the chamber air gives a certain amount of heat products and returned to the collector of solar energy. Products are in a closed form. An additional source of heat in the chamber is exothermic reaction of hydration of cement.

2.2. Heat balances light-transparent cover of collector solar energy

The temperature field in the collector of solar energy during the heating air - nonstationary. The total period during which the heating air is divided into intervals duration $\Delta\tau$. Determination of the maximum period of time $\Delta\tau_{max}$ in accordance with the recommendations given in Chapter 5.

Solar energy supplied to the light-transparent cover of the collector (Fig. 2.2, Fig. 2.3).

Part of solar energy that supplied to the light-transparent cover of the collector for the i-th time interval absorbed light transparent covering part - is reflected, and part passes through the cover:

$$Q_L = Q_A + Q_R + Q_D. \tag{2.1}$$

where Q_L – the amount of solar energy that supplied to the light-transparent cover of the collector for the i-th time interval, J;

Q_A – the amount of solar energy absorbed by the light-transparent cover of the collector for the i-th time interval, J;

$$Q_A = AQ_L\,, \qquad (2.2)$$

A – absorptivity light transparent cover;

Q_R – the amount of solar energy reflected light transparent cover of the collector for the i-th time interval, J;

$$Q_R = RQ_L\,, \qquad (2.3)$$

R – the reflectance of light-transparent cover;

Q_D – amount of solar energy that passes through the light-transparent cover collector for the i-th time interval, J;

$$Q_D = DQ_L, \qquad (2.4)$$

D – the bandwidth light-transparent cover.

Heated light-transparent covering lost certain amount of heat to the environment due to convection. In addition, between light-transparent coating and surfaces arranged around collector solar energy is radiant heat transfer. For quantitative evaluation of said radiation heat transfer is necessary to know the temperature of the heat transfer surfaces (which due to incoming solar energy is constantly changing) their area and location in space. In a real situation quantitatively calculate a radiant heat exchange not possible therefore accepted that the flow resulting radiation between these surfaces is zero ($Q_{RES} = 0$).

The consequence of radiation heat transfer between the light-transparent cover and metal plate is flow resulting radiation.

Heated air moving in collector of solar energy, transfers share heat light transparent covering.

Thermal balance light-transparent coating collector of solar energy for the i-th time interval has the form

$$Q_A + Q_{AL} + Q_{ML} \pm Q_{RES} = Q_{LC} + Q_{LH} + Q_{LV}; \qquad (2.5)$$

given the fact that $Q_{RES} = 0$ (the explanation given above):

$$Q_A + Q_{AL} + Q_{ML} = Q_{LC} + Q_{LH} + Q_{LV}; \qquad (2.6)$$

where Q_A – the same value as in equation (2.1), (2.2);

Q_{AL} – the amount of heat transmitted by the i-th time interval from the air, which moves in the collector solar energy to light transparent cover, J;

Q_{ML} – the amount of heat supplied by the i-th time interval to light transparent cover due to radiation heat transfer between the metal plate and the coating, J;

Q_{LC} – the amount of heat consumed by the i-th time interval for heating the light-transparent cover, J;

Q_{LH}, Q_{LV} – the amount of heat lost by the i-th time interval to the environment through horizontal and vertical structures are light-transparent cover, J.

Schematic reflection of thermal balance (2.6) light-transparent cover collector by the i-th time interval shown in Fig. 2.2.

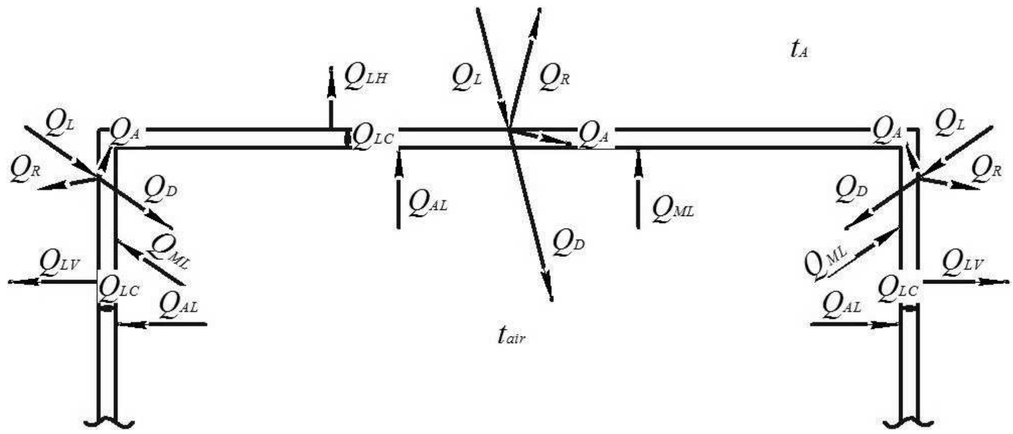


Fig. 2.2. Schematic reflection of thermal balance (2.6)
light-transparent cover the collector for the i-th time interval

If in the certain period of time the temperature of the inner surface of the light-transparent cover higher than the temperature of the air moving in a collector of solar energy, the heat balance of the light-transparent cover has the form

$$Q_A + Q_{ML} = Q_{LA} + Q_{LC} + Q_{LH} + Q_{LV}, \qquad (2.7)$$

where Q_{LA} – the amount of heat that is transmitted over the selected time interval of light-transparent cover to the air moving in a collector of solar energy, J

Schematic reflection of thermal balance (2.7) shown in Fig. 2.3.

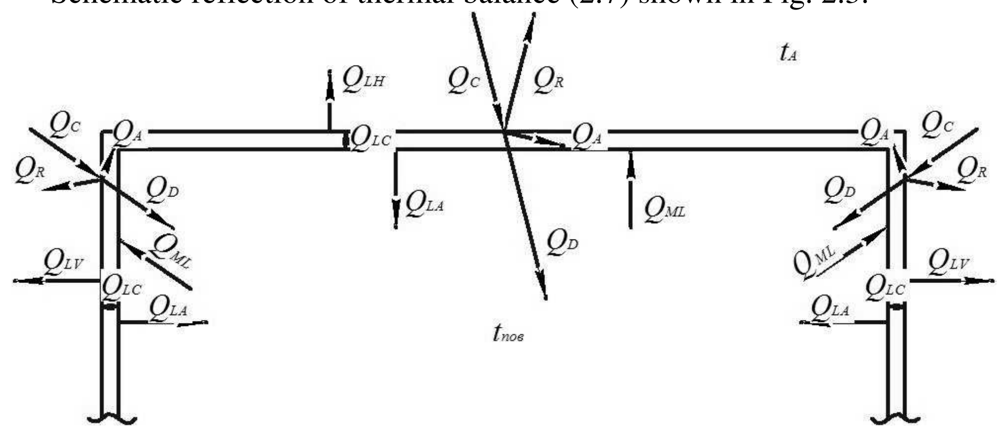


Fig. 2.3. Schematic reflection of thermal balance (2.7)
light-transparent cover of the collector

The amount of heat transmitted by the i-th time interval from the air, which moves in the collector of solar energy to the inner surface of the light-transparent cover, J, calculated by the formula

$$Q_{AL} = \alpha_C \cdot (t_{AIR} - t_{LI}) \cdot F_{LI} \cdot \Delta\tau, \qquad (2.8)$$

accepted simplification: light transparent covering warm up evenly, ie $t_{LI} = t_{AVE}$, then

$$Q_{AL} = \alpha_C \cdot (t_{AIR} - t_{AVE}) \cdot F_{LI} \cdot \Delta\tau, \qquad (2.9)$$

where α_C – heat transfer coefficient of air that moves in the collector of solar energy to the inner surfaces of the light-transparent cover, W/(m^2·ºC);

t_{AIR} – average for the i-th period of time-and temperature in the collector, ºC;

t_{LI} – average for the i-th period of time the temperature inner surfaces of the light-transparent cover, ºC;

t_{AVE} – the average for the i-th period of time temperature light transparent cover, ºC;

F_{LI} – area of internal surfaces of the light-transparent cover, m^2;

$\Delta\tau$ – period of time, s.

Average for the i-th period of time the temperature air in collector of solar energy, ºC, calculated by the equation

$$T_{AIR} = 0{,}5 \cdot (t_{AI} + t_{AO}), \qquad (2.10)$$

where t_{AI}, t_{AO} – the average for the i-th period of time, the air temperature at the inlet to the collector and outlet from it, ºC.

The average for the i-th period of time temperature light transparent covering the collector of solar energy t_A, ºC, is equal to

$$t_A = 0{,}5 \cdot (t_{LB} + t_{LE}), \qquad (2.11)$$

where t_{LB}, t_{LE} – temperature light-transparent cover at the beginning and end of the i-th period of time, ºC.

The amount of heat expended on the heating the light-transparent cover the collector for the i-th period of time J, equals

$$Q_{LC} = c_L \cdot m_L \cdot (t_{LE} - t_{LB}), \qquad (2.12)$$

where c_L – specific mass heat capacity material light-transparent cover, J/(kg·ºC); m_L – mass light-transparent cover, kg.

The amount of heat that is lost by the i-th period of time to the environment through the design of light-transparent cover the collector in the absence of wind, J, calculated by the formulas:

- for the horizontally located constructions light-transparent cover:

$$Q_{LH} = \alpha_{LH} \cdot (t_{LOH} - t_{AMB}) \cdot F_{LOH} \cdot \Delta\tau; \qquad (2.13)$$

considering accepted before simplification (light-transparent cover the collector to warm up evenly)

$$Q_{LH} = \alpha_{LH} \cdot (t_A - t_{AMB}) \cdot F_{LOH} \cdot \Delta\tau; \qquad (2.14)$$

- for vertically located constructions light-transparent cover:

$$Q_{LV} = \alpha_{LV} \cdot (t_{LOV} - t_{AMB}) \cdot F_{LOV} \cdot \Delta\tau, \qquad (2.15)$$

considering accepted before simplification (light-transparent coating evenly-heated)

$$Q_{LV} = \alpha_{LV} \cdot (t_A - t_{AMB}) \cdot F_{LOV} \cdot \Delta\tau, \qquad (2.16)$$

where α_{LH} – heat transfer coefficient from the outer surface of the horizontal constructions light-transparent cover the collector to the environment, W/(m^2·ºC);

α_{LV} – heat transfer coefficient of the external surfaces of the vertical constructions of light-transparent cover the collector of solar energy to the environment, W/(m^2·ºC);

t_{LOH} – the average for the i-th period of time temperature of the outer surface of the horizontally located constructions light-transparent cover the collector of solar energy, ºC;

t_{LOV} – the average for the i-th period of time temperature of the outer surface vertically located constructions of light-transparent cover the collector, ºC;

t_A – the average for the i-th period of time temperature light transparent cover the collector, ºC;

t_{AMB} – ambient temperature, ºC;

F_{LOH}, F_{LOV} – area horizontally and vertically located external surfaces of the light-transparent cover the collector, m^2;

$\Delta\tau$ – period of time, s.

Subject to availability of wind $\alpha_{CB} = \alpha_{C\Gamma} = \alpha_{C3}$, then

$$Q_{C\Gamma} + Q_{CB} = \alpha_{C3}\cdot(t_C - t_{HC})\cdot(F_{C3\Gamma} + F_{C3B})\cdot\Delta\tau. \qquad (2.17)$$

2.3. Heat balance heat-accepting metal plate and layer of thermal insulation collector of solar power for i-th period of time

Thermal balance metal plate and layer of thermal insulation collector of solar energy for i-th period of time has the form

$$Q_D = Q_M + Q_{MA} + Q_{MLC} + Q_{TI} + Q_{TIH}, \qquad (2.18)$$

where Q_D – the same value as in equation (2.1), (2.4);

Q_M – the amount of heat it takes to heat a metal plate for i-th period of time J;

Q_{MA} – the amount of heat transmitted for i-th period of time from the metal plate to the air, which moves in the collector, J;

Q_{MLC} – the amount of heat transferred from the surface of the metal plate to the inner surface of the light-transparent cover for i-th period of time because of radiant heat transfer between them, J;

Q_{TI} – the amount of heat expended on heat of thermal insulation layer for i-th period of time J;

Q_{TIH} – the amount of heat lost for i-th period of time to the environment through a horizontal surface layer thermal insulation, J (subject to the availability of heat losses in the i-th period of time).

Schematic reflection of thermal balance (2.18) is shown in Fig. 2.4.

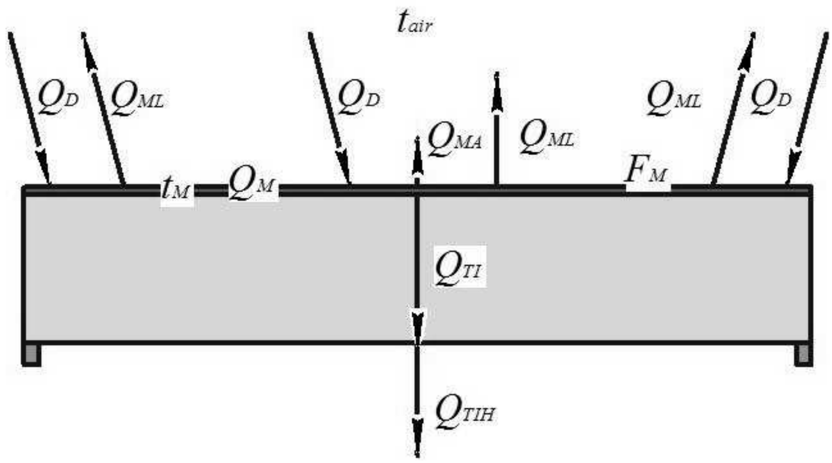


Fig. 2.4. Schematic reflection of thermal balance (2.18) heat-accepting metal coating and of thermal insulation layer solar energy collector

If the use of solar energy collector is during the spring period, it is advisable to consider: the heat loss to the environment through the vertical surface layer of thermal insulation; the cost of heat for heating bearing support; heat loss to the environment bearing support.

For spring and autumn the heat balance of the metal plate and layer of thermal insulation collector of solar energy for i-th period of time has the form

$$Q_D = Q_M + Q_{MA} + Q_{MLC} + Q_{TI} + Q_{TIH} + Q_{TIV} + Q_S + Q_{SE}, \quad (2.19)$$

where Q_{TIV} – the amount of heat lost for i-th period of time to the environment through the vertical surface layer of thermal insulation, J;

Q_S – the amount of heat consumed for the i-th period of time for heating the supports, J (subject to the availability of heat losses in the i-th period of time);

Q_{SE} – the amount of heat lost for the i-th period of time to the environment from the supports, J. (subject to the availability of heat losses in the i-th period of time).

Schematic reflection of thermal balance (2.19) is shown in Fig. 2.5.

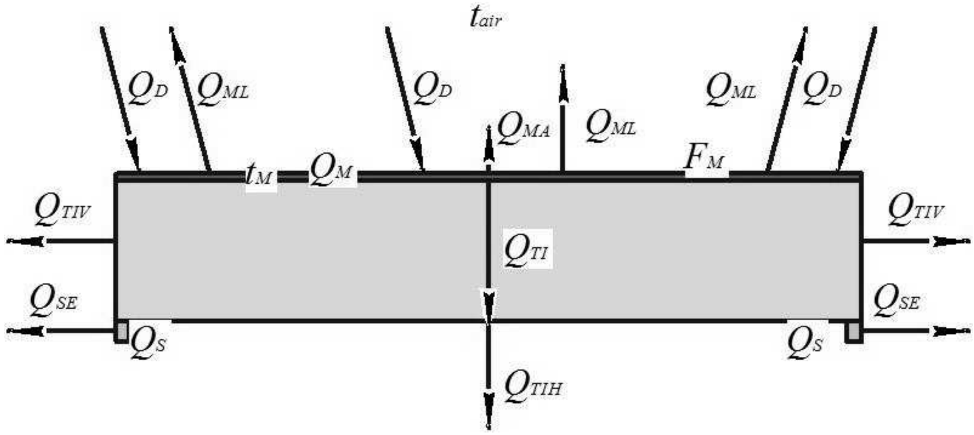


Fig. 2.5. Schematic reflection of thermal balance (2.19) heat-accepting metal coating and of thermal insulation layer solar energy collector

The amount of heat expended on heating a metal plate on the i-th period of time J; calculated by dependence

$$Q_M = c_M \cdot m_M \cdot (t_{ME} - t_{MB}), \tag{2.20}$$

where c_M – specific heat capacity mass metal plate, J/(kg·°C);

m_M – weight metal plate kg, which is calculated by the equation

$$m_M = \rho_M \cdot V_M, \tag{2.21}$$

ρ_M – density of metal, kg/m^3; V_M – volume of metal plate, m^3;

t_{MB}, t_{ME} – the temperature of the metal plate at the beginning and end of the i-th period of time, °C.

The amount of heat transmitted by the i-th period of time from the metal plate to the air, which moves in the collector, J, calculated by the formula

$$Q_{MB} = \alpha_C \cdot (t_{MO} - t_{AIR}) \cdot F_M \cdot \Delta\tau; \tag{2.22}$$

accepted that the metal plate has a relatively small thickness, and therefore warmed uniformly, ie,, $t_{MO} = t_M$, then

$$Q_{MB} = \alpha_C \cdot (t_M - t_{AIR}) \cdot F_M \cdot \Delta\tau, \tag{2.23}$$

where α_C – coefficient of heat transfer from the metal plate to the air, W/(m^2·°C);

t_{MO} – and the average for the i-th period of time temperature of the outer surface of the metal plate, °C;

t_M – and the average for the i-th period of time temperature metal plate, °C;

t_{AIR} – and the average for the i-th period of time temperature in the collector, °C;

F_M – area of metal plate, m^2; $\Delta\tau$ – period of time, s.

Average for the i-th period of time temperature metal plate t_M is equal to

$$t_M = 0{,}5 \cdot (t_{MB} + t_{ME}). \tag{2.24}$$

The amount of heat Q_{MLC}, transmitted by the i-th period of time from the metal plate to the light-transparent cover due to the presence of radiation heat transfer between them, J, is calculated by the formula given in sources [27], [28],

$$Q_{MLC} = \varepsilon_{п} \cdot c_0 \cdot \left[\left[\left(\frac{T_{MO}}{100}\right)\right]^4 - \left[\left(\frac{T_{LI}}{100}\right)\right]^4\right] \cdot F_M \cdot \Delta\tau \,, \qquad (2.25)$$

taking into account previously adopted simplifications ($T_{LI} = T_L$, $T_{MO} = T_M$):

$$Q_{MLC} = \varepsilon_{п} \cdot c_0 \cdot \left[\left[\left(\frac{T_M}{100}\right)\right]^4 - \left[\left(\frac{T_L}{100}\right)\right]^4\right] \cdot F_M \cdot \Delta\tau \,, \qquad (2.26)$$

where T_{MO} – the average for the i-th period of time the absolute temperature of the outer surface of the metal plate, K;

T_M – average for the i-th period, and the absolute temperature of the metal plate, K;

T_{LI} – the average for the i-th period of time absolute temperature of the inner surfaces of the light-transparent cover, K;

T_L – the average for the i-th period of time absolute temperature light-transparent cover, K;

c_0 – radiation coefficient absolute black body, $W/(m^2 \cdot K^4)$;

$\Delta\tau$ – period of time, s;

ε_s – shows the degree of blackness, which is calculated by the formula

$$\varepsilon_s = \frac{1}{\frac{1}{\varepsilon_M} + \frac{F_M}{F_{LI}} \cdot \left(\frac{1}{\varepsilon_L} - 1\right)} \,, \qquad (2.27)$$

where ε_M – degree of blackness metal plate;

ε_L – degree of blackness light-transparent cover;

F_M – area of metal plate, m^2;

F_{LI} – area of of internal surfaces of the light-transparent cover, m^2;.

Methods for calculating the amount of heat Q_H, expended on heating of thermal insulation layer on the i-th period of time, see in chapter 5.

The amount of heat that is lost by the i-th period of time to the environment through a layer of of thermal insulation in the absence of wind, J, is calculated by the formula:

- for horizontally located surface:

$$Q_{TIH} = \alpha_{TIH} \cdot (t_{TIH} - t_{AMB}) \cdot F_{TIH} \cdot \Delta\tau \,; \qquad (2.28)$$

- for vertically located surfaces:

$$Q_{TIV} = \alpha_{TIV} \cdot (t_{TIV} - t_{AMB}) \cdot F_{TIV} \cdot \Delta\tau, \qquad (2.29)$$

where α_{TIH}, α_{TIV} – heat transfer coefficient from horizontal and vertical located outside surfaces layer of thermal insulation to the environment, $W/(m^2 \cdot ^\circ C)$; the resulting emission flow between these surfaces and surfaces that surround the solar energy collector, made to be zero, then the coefficients α_{TIH}, α_{TIV} reflecting the intensity of convective heat transfer;

t_{TIH}, t_{TIV} – average of the i-th period of time temperature external surfaces horizontally and vertically located surface layer of thermal insulation, ºC;

t_{AMB} – ambient temperature, ºC;

F_{TIH}, F_{TIV} – area of horizontally and vertically located surface layer of thermal insulation, m^2; $\Delta\tau$ – period of time, s.

Subject to availability of wind $\alpha_{TIH} = \alpha_{TIV} = \alpha_{TI}$.

2.4. Thermal balance collector of solar energy for the i-th period of time

2.4. Thermal balance collector of solar energy for the i-th period of time has the form

$$Q_A + Q_D + Q_E = Q_M + Q_R + Q_{LC} + Q_{LH} + Q_{LV} + Q_H + Q_{TIH}, \qquad (2.30)$$

where Q_E – the amount of heat that enters the solar energy collector with air for the i-th period of time, J;

Q_R – the amount of heat removed from the solar energy collector with air for the i-th period of time, J;

Explanation of other components of this thermal balance above.

Schematic reflection components of thermal balance (2.30) are shown in Fig. 2.6.

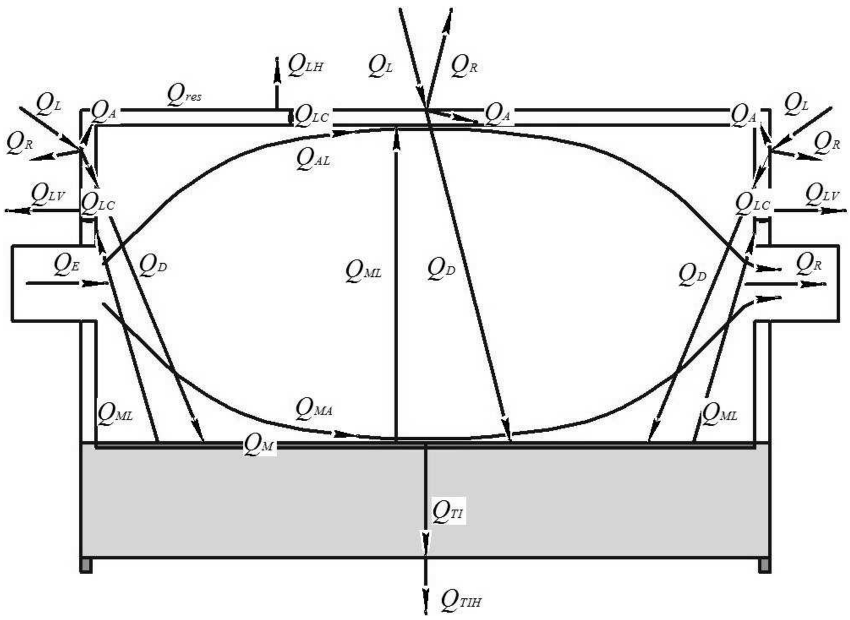


Fig. 2.6. Schematic reflection thermal balance collectors of solar energy (2.30)

For spring and autumn period thermal balance collector of solar energy for the i-th period of time has the form

$$Q_A + Q_D + Q_E = Q_M + Q_R + Q_{LC} + Q_{LH} + Q_{LV} + Q_H + \\ + Q_{TIH} + Q_{TIV} + Q_S + Q_{SE}, \quad (2.31)$$

Schematic reflection components of thermal balance (2.31) are shown in Fig. 2.7.

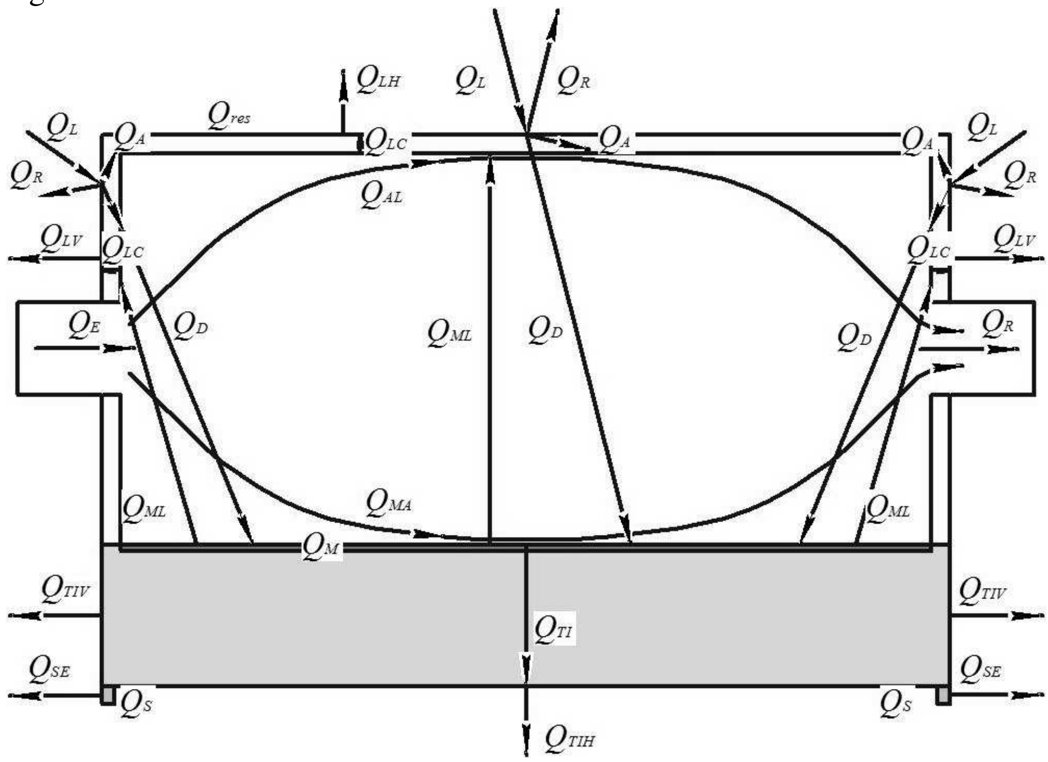


Fig. 2.7. Schematic reflection thermal balance collector of solar energy (2.31)

The amount of heat, J, entering to the collector of solar energy with air for the i-th period of time is calculated by dependence

$$Q_I = c_{AI} \cdot \rho_{AI} \cdot L \cdot t_{AI} \cdot \Delta\tau, \quad (2.32)$$

where t_{AI} – the average for the i-th period of time, the air temperature at the inlet to the collector, ºC;

c_{AI} – specific mass heat capacity of air at temperature $t_{\Pi H}$, J/(kg·ºC);

ρ_{AI} – air density at the temperature t_{AIR}, kg/m^3;

L – air consumption, m^3/s; $\Delta\tau$ – period of time, s.

The amount of heat, J, remove from the solar energy collector with air at the i-th period of time determined by the dependence

$$Q_O = c_{AO} \cdot \rho_{AO} \cdot L \cdot t_{AO} \cdot \Delta\tau, \quad (2.33)$$

where t_{AO} – average for the i-th period of time, the air temperature at the outlet of the collector, ºC;

c_{AO} – specific mass heat capacity of air at the temperature t_{AO}, J/(kg·ºC);

ρ_{AO} – air density at the temperature t_{AO}, kg/m^3.

Chapter 3. Definition of temperature change of concrete products and changes in air temperature in the thermal chamber

We consider the heat exchange in the chamber intended for thermal treatment hot air of concrete paving slabs. The tiles are placed on the shelves.

Products heating due to intake in equipment heated in collector of solar energy air and due to the presence of air and heat isolation hydration of cement.

Heated air moving in the chamber should not face the open surface of the product that there is no evaporation of moisture required for hydration of cement. We consider three variants waterproofing products.

It is assumed that in an hydro-isolated system thickness of air layer is minimal, so you can ignore the presence of: mass exchange processes between the exposed surface of products and air layers; the influence of the air layer on the processes of heat transfer in hydro-isolated system.

The total period for which the product is heated, divided into periods of time duration $\Delta\tau$. Determination of the maximum period of time $\Delta\tau_{max}$ in accordance with the recommendations given in chapter 5.

3.1. Determination of the temperature change of concrete products and changes in air temperature in heating chamber (variant 1)

The scheme first version waterproofing of concrete products (paving slabs) is shown in Fig. 3.1.

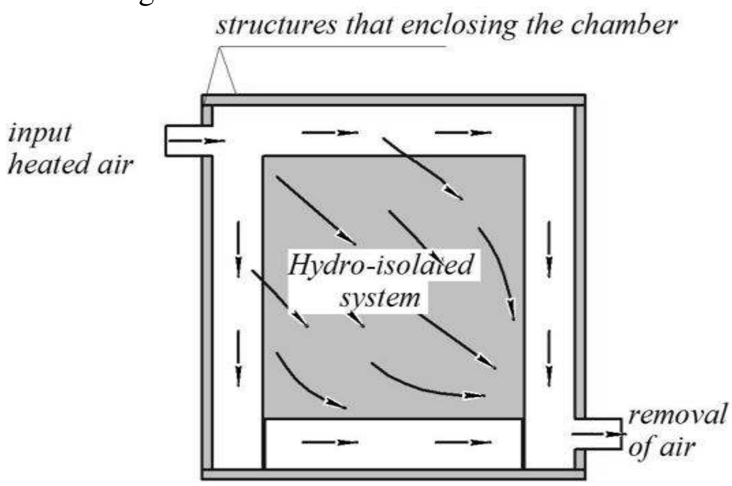


Fig. 3.1. The scheme first version waterproofing of concrete products

Thermal balance of the chamber for thermal treatment of concrete products hot air for the i-th period of time has the form

$$Q_{CE} + Q_{EXO} = Q_{CR} + Q_{HI} + Q_{CW} + Q_{CO} + Q_{FL} + Q_{WA} + Q_{OA} + Q_{S} + Q_{E}, \quad (3.1)$$

where Q_{CE} – the amount of heat that enters the chamber and during the i-th period of time with the heated air, J;

Q_{EXO} – heat-revenues during the i-th period of time due to the presence of exothermic reaction of hydration of cement, J;

Q_{CR} – heat loss during the i-th period of time with the air removed from the chamber, J;

Q_{HI} – the loss of heat for heating components of hydro-isolated system for the i-th period of time J;

Q_{CW} – the loss of heat for heating the walls of chamber for the i-th period of time J;

Q_{CO} – the loss of heat for heating overlapping camera for the i-th period of time J;

Q_{FL}– the loss of heat for heating floors camera for the i-th period of time J;

Q_{WA} – heat loss to the environment through walls camera for i-th period of time J, (subject to the availability of these losses in the i-th period of time);

Q_{OA} – heat loss to the environment through the overlapping camera for the i-th period of time J. (subject to the availability of these losses in the i-th period of time);

Q_{S} – the loss of heat for the i-th period of time for heating the soil, J. (за subject to the availability of these losses in the i-th period of time);

Q_{E} – the loss of heat for the i-th period of time for heating equipment chamber (supports hydro-isolated systems, air pipes), J.

Fig. 3.2 schematically shows the components of the thermal balance of the camera (variant 1).

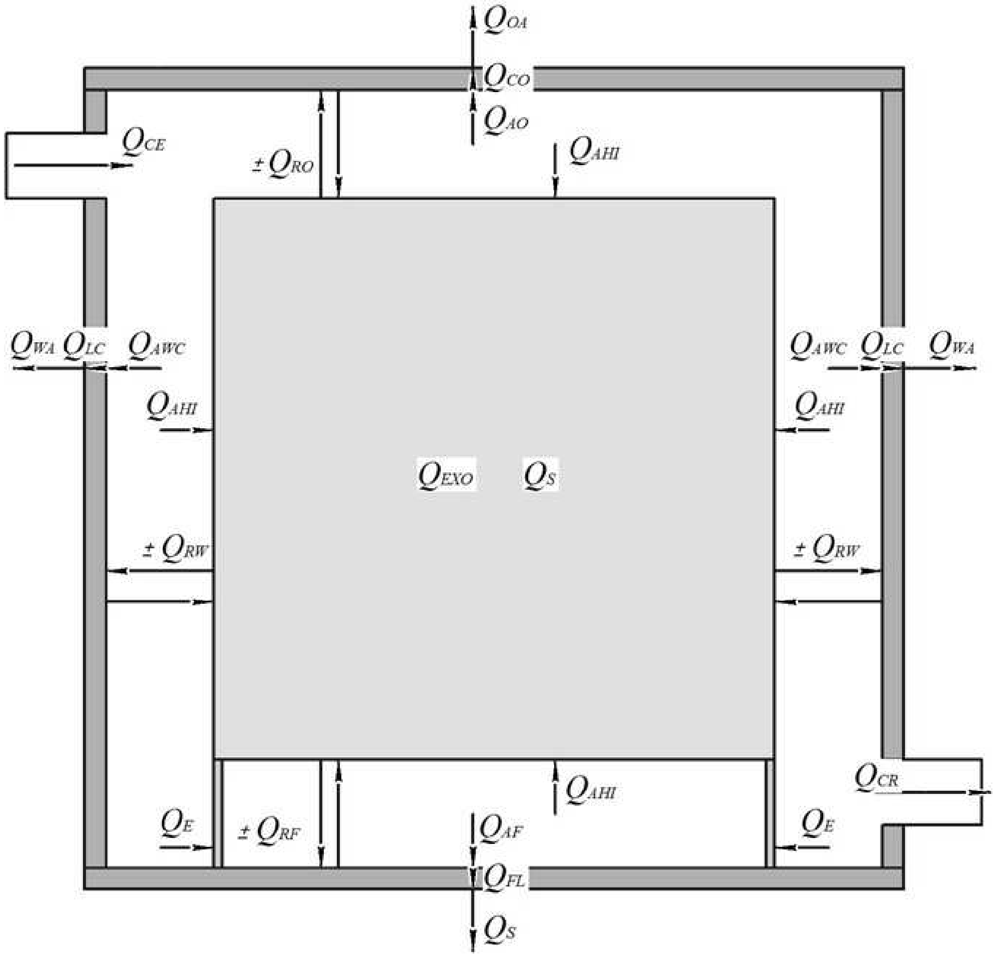


Fig. 3.2. Schematic reflection components heat balances camera (variant 1)

The amount of heat Q_{CE}, entering the chamber during the i-th period of time with heated air, J, calculated by dependence

$$Q_{CE} = c_{AIR} \cdot \rho_{AIR} \cdot L \cdot t_{AIR} \cdot \Delta\tau, \quad (3.2)$$

where c_{AIR} – specific mass heat capacity of air entering into the chamber, J/(kg·ºC); t_{AIR} – average for the i-th period of time temperature of air entering into the chamber, ºC; L – air consumption, m^3/c; $\Delta\tau$ – period of time, c.

The amount of heat Q_E, and released during the i-th period of time due to hydration of cement is determined by the experimental data and other things being equal depends on the temperature of the concrete.

The heat loss Q_{CR} during the i-th period of time with the air is removed from the camera, J, is calculated for dependence

$$Q_{CR} = c_{AE} \cdot \rho_{AE} \cdot L \cdot t_{AE} \cdot \Delta\tau, \quad (3.3)$$

where c_{AE} – specific mass heat capacity air removed from the camera, J/(kg·ºC);

t_{AE} – середня for the i-th period of time temperature of air is removed from the camera, ºC;

ρ_{AE} – air density at the temperature t_{AE}, kg/m^3.

The heat loss Q_{HI} for heating the component hydro-isolated system for the the i-th period of time equal

$$Q_{HI} = Q_C + Q_F + Q_{HIE} + Q_{HIM}, \quad (3.4)$$

where Q_C, Q_F, Q_{HIE}, Q_{HIM} – the loss of heat for heating concrete products, molds, equipment hydro-isolated systems, waterproofing material for the i-th period of time J, if the equipment hydro-isolated system made of various materials, this factor is taken into account in the calculations.

The heat loss Q_{HI} for heating the component hydro-isolated system for the i-th period of time, can be determined by dependence

$$Q_{HI} = c_{HI} \cdot m_{HI} \cdot (t_{HIE} - t_{HIS}), \quad (3.5)$$

where c_{HI} – average specific mass heat capacity materials hydro-isolated system, J/(kg·ºC);

m_{HI} – weight components of hydro-isolated system, kg;

t_{HIS} – the average temperature of the components of hydro-isolated system at the start the i-th period of time, ºC;

t_{HIE} – the average temperature of the components of hydro-isolated system the end of the i-th period of time, ºC.

Changing the temperature of components of hydro-isolated system is due to the presence of:

- heat flow directed from air to surface hydro-isolated system;
- exothermic reaction of hydration of cement;
- flow of the resulting radiation between the surface hydro-isolated system and the internal surfaces of chamber; flow direction resulting radiation depends on the ratio between the surface temperature hydro-isolated system and temperature internal surfaces of structures enclosing chamber.

That is, the loss of heat Q_{HI} for heating the component hydro-isolated system for the i-th period of time equal

$$Q_{HI} = c_{HI} \cdot m_{HI} \cdot (t_{HIE} - t_{HIS}) = Q_{AHI} + Q_{EXO} \pm Q_{RW} \pm Q_{RO} \pm Q_{RF}, \quad (3.6)$$

where Q_{AHI} – the amount of heat which perceives hydro-isolated system of heated air for the i-th period of time J;

Q_{EXO} – heat-revenues during the i-th period of time due to the presence of exothermic reaction of cement hydration, J;

Q_{RW} – the amount of heat J, transmitted for i-th period of time due to the presence of resulting radiation from vertical surfaces hydro-isolated system to the internal wall surfaces camera (or in the reverse direction); flow direction resulting radiation depends on the ratio between the temperatures of these surfaces;

Q_{RO} – amount of heat, J, transmitted by the i-th period of time due to the presence of resulting radiation from the inverse up horizontal surface hydro-isolated system to the inner surface of overlapping cameras (or in the reverse direction); flow direction resulting radiation depends on the ratio between the temperatures of these surfaces;

Q_{RF} – amount of heat, J, transmitted by the i-th period of time due to the presence of resultingradiation from the inverse down horizontal surface hydro-isolated system to the floor camera (or in the reverse direction); flow direction resulting radiation depends on the ratio between the temperatures of these surfaces.

The components of (3.6) are interrelated, so identifying with this thermal balance temperature change of concrete products for the i-th period of time taken simplification.

In reference literature (particularly in the source [29]) are the following dependence to determine the amount of heat Q, J, which takes material from the air over time:

$$Q = c \cdot m \cdot (t_{at} - t_{Mi}) \cdot \beta, \tag{3.7}$$

where c – specific mass heat capacity material, J/(kg·ºC);

m – the mass of material, kg;

t_{at} – the ambient temperature, ºC;

t_{Mi} – the initial temperature of the material, ºC;

β – the coefficient, which takes into account the proportion of heat treats the material for the selected period of time (relative to the amount of heat required to heating the material to ambient temperature); the coefficient β determined by the reference data by using Fourier criterion of equal

$$F_O = \Delta\tau \ / c \cdot m \cdot R, \tag{3.8}$$

where Δτ – period of time, s;

R – full resistance transfer to heat from the environment to the heat transfer surface, ºC/W.

The value R is calculated by the formula

$$R = m / (\rho \cdot \lambda \cdot F^2) + 1 / (\alpha \cdot F), \tag{3.9}$$

where ρ – density material, kg/m^3;

λ – the coefficient of thermal conductivity material, W/(m·ºC);

α – coefficient of heat transfer from the environment to the heat transfer surface, W/(m^2·ºC);

F – heat transfer surface area, m^2.

For use the above formulas offered to apply simplification: the value c_{HI}, ρ_{HI} and λ_{HI} equal:

$$c_{HI} = (m_C \cdot c_C + m_F \cdot c_F + m_{HIE} \cdot c_{HIE} + m_{HIM} \cdot c_{HIM}) \ / \ m_{HI}; \tag{3.10}$$

$$\lambda_{HI} = (m_C \cdot \lambda_C + m_F \cdot \lambda_F + m_{HIE} \cdot \lambda_{HIE} + m_{HIM} \cdot \lambda_{HIM}) \ / \ m_{HI}; \tag{3.11}$$

$$\rho_{HI} = (m_C \cdot \rho_C + m_F \cdot \rho_F + m_{HIE} \cdot \rho_{HIE} + m_{HIM} \cdot \rho_{HIM}) \ / \ m_{HI}, \tag{3.12}$$

where c_{HI}, ρ_{HI}, λ_{HI} – respectively average specific mass heat capacity, average density and average coefficient of thermal conductivity materials hydro-isolated system;

c_C, c_F, c_{HIE}, c_{HIM} – Specific mass heat capacity under concrete material forms, material equipment of hydro-isolated systems and hydro-insulation material, J/(kg·ºC);

λ_C, λ_F, λ_{HIE}, λ_{HIM} – coefficient of thermal conductivity under concrete, material form, material equipment of hydro-isolated systems and hydro-insulation material, W/(m·ºC);

ρ_C, ρ_F, ρ_{HIE}, ρ_{HIM} – respectively density concrete, material forms, material equipment of hydro-isolated systems and hydro-insulation material, kg/m^3;

m_{HI} – mass hydro-isolated system m_{HI}, kg, equal

$$m_{HI} = m_C + m_F + m_{HIE} + m_{HIM}, \tag{3.13}$$

where m_C, m_F, m_{HIE}, m_{HIM} – mass respectively concrete mix (concrete), forms, equipment hydro-isolated system, hydro-insulation material kg.

Given these simplifications

$$F_O = \Delta\tau / c_{HI}\cdot m_{HI}\cdot R ; \tag{3.14}$$

$$R = m_{HI} / (\rho_{HI}\cdot\lambda_{HI}\cdot F_{HI}^{\ 2}) + 1 / (\alpha\cdot F_{HI}), \tag{3.15}$$

where F_{HI} – Surface area of hydro-isolated system, m^2;

α – coefficient of heat transfer from the heated air to the heat transfer surfaces camera, W/(m^2·ºC).

Then *the amount of heat* Q_{AHI}, J, which perceives hydro-isolated system of heated air for the i-th period of time, is equal to:

$$Q_{AHI} = c_{HI}\cdot m_{HI}\cdot(t_{AIR} - t_{HIS})\cdot\beta, \tag{3.16}$$

where t_{AIR} – average for the i-th period of time temperature in the chamber, ºC;

t_{HIS} – average temperature hydro-isolated system at the start the i-th period of time, ºC.

Average for the i-th period of time the air temperature in the chamber, ºC, calculated by the equation

$$T_{AIR} = 0{,}5\cdot(t_{AE} + t_{AR}), \tag{3.17}$$

where t_{AE} – average for the i-th period of time temperature of air entering the camera, ºC;

t_{AR} – average for the i-th period of time temperature, removed from the camera, ºC.

The amount of heat, J, is transmitted by the i-th period of time due to the presence of the resulting radiation Q_{RW}, Q_{RO}, Q_{RF}, calculated by the general formula given in particular in the sources [27], [28]:

$$Q_{12} = \varepsilon_s\cdot c_0\cdot\left[\left[\left(\frac{T_1}{100}\right)\right]^4 - \left[\left(\frac{T_2}{100}\right)\right]^4\right]\cdot F_1\cdot\Delta\tau. \tag{3.18}$$

where c_0 – coefficient of radiation of a blackbody, W/(m^2 ·K^4);

T_1, T_2 – temperature heat transfer surfaces, m^2;

$\Delta\tau$ – period of time, s;

ε_s – shows the degree of blackness, which is calculated by the formula

$$\varepsilon_s = \frac{1}{\frac{1}{\varepsilon_1} + \frac{F_1}{F_2} \cdot \left(\frac{1}{\varepsilon_2} - 1\right)}, \tag{3.19}$$

where F_1, F_2 – heat transfer surface area, m^2.

The loss of heat for heating the chamber walls Q_{CW}, on the heating overlapping chamber Q_{CO} and floor heating chamber Q_{FL} for the i-th period of time calculated by the dependencies, presented in Chapter 5. The basis for the use of these dependencies is thermal balances constructions enclosing chamber.

Thermal balance walls chambers for the i-th period of time has the form

– provided that the temperature vertical surfaces hydro-isolated system is higher than the temperature of the internal surfaces of the chamber walls:

$$Q_{AWC} + Q_{RW} = Q_{CW} + Q_{WE}, \tag{3.20}$$

– provided that the temperature of the internal surfaces of the chamber walls higher than the temperature vertical surfaces hydro-isolated system:

$$Q_{AWC} = Q_{CW} + Q_{WE} + Q_{RW}, \tag{3.21}$$

where Q_{AWC} – the amount of heat transferred from the heated air to the walls of the chamber for the i-th period of time J;

Q_{CW} – the loss of heat for heating the walls of the chamber for the i-th period of time J;

Q_{WE} – heat loss into the environment through walls camera for i-th period of time, J, (provided these losses in the i-th period of time);

Q_{RW} – the same value that in (3.6).

Thermal balance overlapping camera for the i-th period of time is:

- Provided that the temperature inverse up horizontal surface hydro-isolated system is higher than the temperature of the inner surface overlapping camera:

$$Q_{AO} + Q_{RO} = Q_{CO} + Q_{OE}, \tag{3.22}$$

– provided that the temperature of the inner surface of the overlapping chamber is higher than the temperature inverse up horizontal surface hydro-isolated system:

$$Q_{AO} = Q_{CO} + Q_{OE} + Q_{RO}, \tag{3.23}$$

where Q_{AO} – the amount of heat transferred from the heated air to overlapping camera for i-th period of time, J;

Q_{CO} – the loss of heat for heating overlapping camera for i-th period of time J;

Q_{OE} – heat loss into the environment through the overlapping camera for i-th period of time J;

Q_{RO} – the same value that in (3.6).

Thermal balance of the floors camera for the i-th period of time is:

- provided that the temperature inverse down horizontal surface hydro-isolated system is higher than the temperature of the floor camera:

$$Q_{AF} + Q_{RF} = Q_{FL} + Q_S , \tag{3.24}$$

– provided that the surface temperature of the floor chamber is higher than the temperature inverse down horizontal surface hydro-isolated system:

$$Q_{AF} = Q_{FL} + Q_S + Q_{RF}, \tag{3.25}$$

where Q_{AF} – the amount of heat is transmitted by the i-th period of time from the heated air to the floor chamber, J;

Q_{FL}– the loss of heat for the i-th period of time for heating floor chamber, J;

$Q_{ГP}$ – the loss of heat for the i-th period of time for heating the soil, J (provided these losses in the i-th period of time);

an explanation of Q_{RF} given the formula (3.6).

The heat loss Q_{WE} into the environment through walls camera for i-th period of time, J, is calculated using the formula

$$Q_{WE} = \alpha_{WE} \cdot (t_{WO} - t_E) \cdot F_{WO} \cdot \Delta\tau , \tag{3.26}$$

where α_{WE} – heat transfer coefficient of the external surfaces of the walls of the chamber to the environment, W/(m^2·ºC);

t_{WO} – the average for the selected period of time the outer surface temperature of the chamber walls, ºC;

F_{WO} – the area of the outer surface of the chamber walls, m^2.

The heat loss Q_{OE} into the environment through the overlapping camera for i-th period of time, J, is calculated using the formula

$$Q_{OE} = \alpha_{OE} \cdot (t_{OO} - t_E) \cdot F_{OO} \cdot \Delta\tau , \tag{3.27}$$

where α_{OE} – coefficient of of heat transfer from the outer surface of the overlapping camera to the environment, W/(m^2·ºC);

t_{OO} – average for the selected period of time the outer surface temperature of overlapping cameras, ºC;

F_{OO} – external surface area of overlapping cameras, m^2.

3.2. Determination of the temperature change of concrete products and changes in air temperature in heating chamber (variant 2)

The second variant waterproofing of concrete products (paving slabs) is shown in Fig. 3.3.

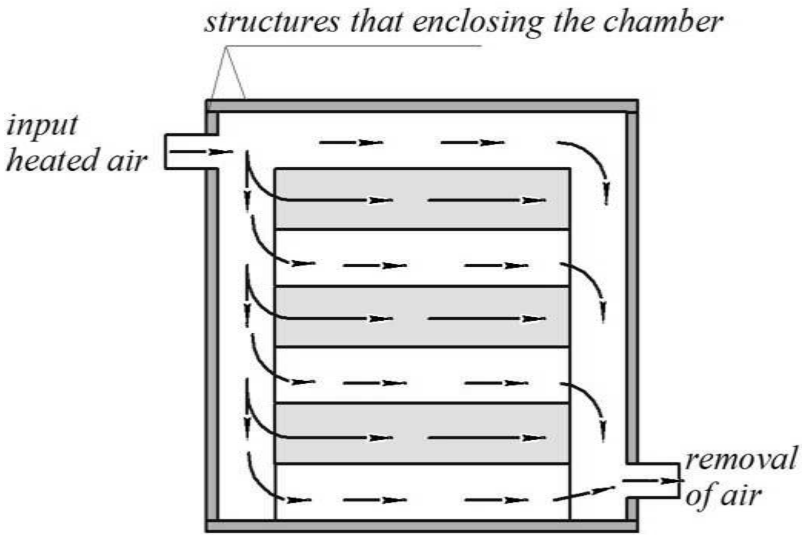


Fig. 3.3. The second variant waterproofing of concrete products

Thermal balance chamber (variant 2) for the thermal treatment of concrete products hot air for the i-th period of time has the form

$Q_{CE} + Q_{EXO} = Q_{CR} + Q_{HIB} + Q_{HI} + Q_{CW} + Q_{CO} + Q_{FL} + Q_{WA} + Q_{OA} + Q_S + Q_E$, (3.28)

where Q_{CE}, Q_{EXO}, Q_{CR},Q_{HI} ,Q_{CW} ,Q_{CO} ,Q_{FL} , Q_{WA} , Q_{OA} , Q_S ,Q_E, – the same the value as in the thermal balance (3.1), the calculation takes into account their design features of the camera (variant 2);

Q_{HIB} – the loss of heat for heating hydro-insulated products for the i-th period of time, J.

If the camera simultaneously carried out thermal treatment of concrete paving slabs several standard sizes, the temperature of the components of the upper, intermediate and lower block products will be different, which is taken into account in the selection below heat balance.

Thermal balance upper hydro-insulated products for the i-th period of time has the form

$Q_{BU} = c_{BU} \cdot m_{BU} \cdot (t_{BUE} - t_{BUB}) = Q_{ABU} + Q_{EU} \pm Q_{RWU} \pm Q_{ROU} \pm Q_{RU}$, (3.29)

where Q_{BU} – the loss of heat for heating the upper block products for the i-th period of time J;

c_{BU} – average specific mass heat capacity materials upper hydro-insulated block products, J/(kg·ºC);

m_{BU} – mass components upper of hydro-insulated block products, kg;

t_{BUB}, t_{BUE} – the average temperature of components of the upper block of the products at the beginning and end of the i-th period of time, ºC;

Q_{ABU} – the amount of heat that perceives the upper block products from heated air for the i-th period of time, J;

Q_{EU} – heat-receipts during for the i-th period of time due to the presence of exothermic reaction of hydration of cement in the upper block products, J;

Q_{RWU} – the amount of heat J, transmitted for i-th period of time due to the presence of resulting radiation from vertical surfaces of the upper block products to the relevant internal surfaces of the walls of the chamber or in the opposite direction;

Q_{ROU} – the amount of heat J, transmitted for i-th period of time due to the presence of resulting radiation from the inverse up horizontal surface of the upper block products to the inner surface of the overlapping chamber or in the opposite direction;

Q_{RU} – the amount of heat J, transmitted for the i-th period of time due to the presence of resulting radiation from the inverse down horizontal surface the upper block of products to the inverse up horizontal surface located below block product or backwards.

The direction of these flows resulting radiation depends on the ratio between the temperatures corresponding surfaces.

Fig. 3.4. schematically shows the components of the heat balance of the camera (variant 2).

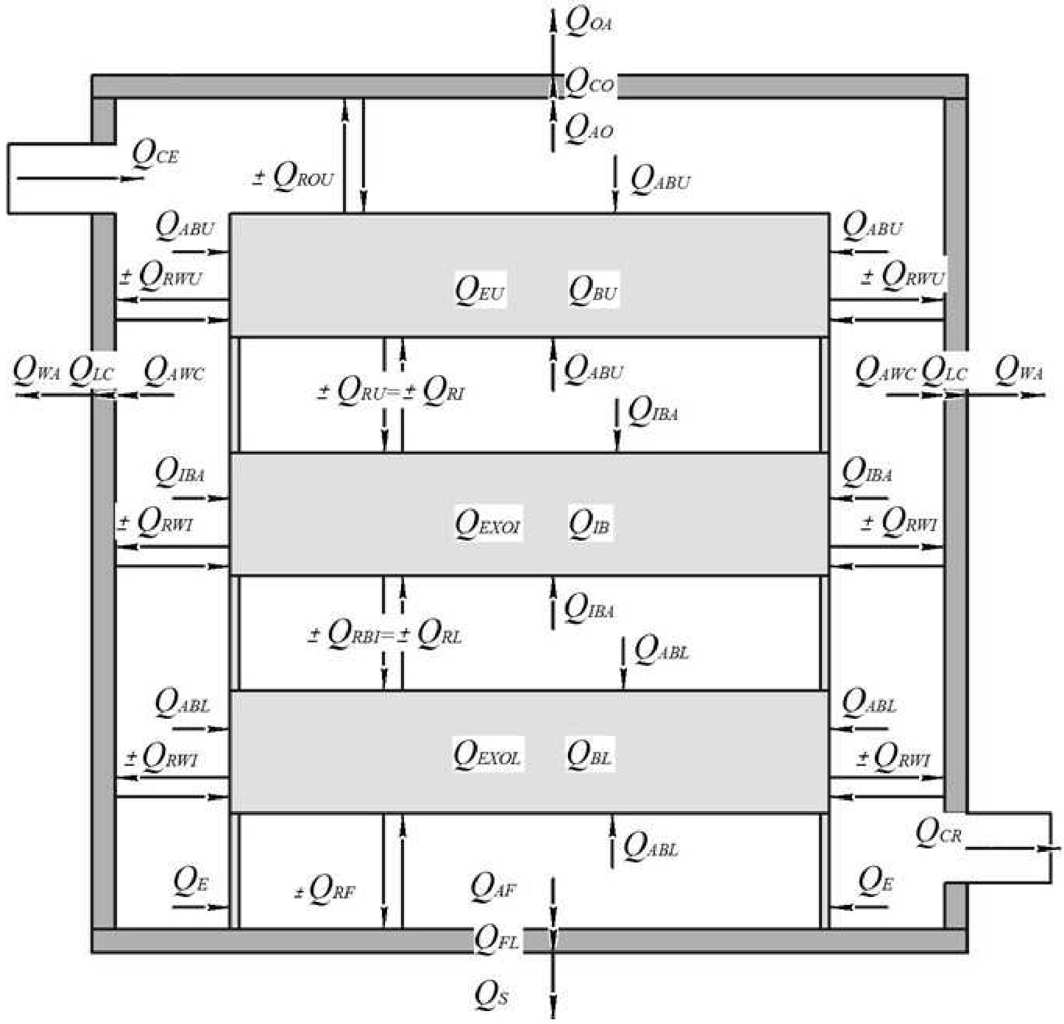


Fig. 3.4. Schematic reflection components heat balances camera (variant 2)

The components of (3.29) are interconnected, so the definitions through this heat balance temperature changes of concrete products for the i-th period of time taken simplification.

The amount of heat Q_{ABU}, which perceives the upper hydro-isolated block products from hot air for the i-th period of time J, is equal to

$$Q_{ABU} = c_{BU} \cdot m_{BU} \cdot (t_{П} - t_{BUB}) \cdot \beta_U, \qquad (3.30)$$

where $t_{П}$ – average for the i-th period of time temperature in the chamber, ºC;

t_{BUB} – the average temperature of the components upper block of products at the beginning and the i-th period of time, ºC;

β_U – coefficient of, which takes into account the proportion of heat perceives the upper block products for the i-th period of time (relative to the amount of heat needed for the heating block to a temperature t_{AIR}).

Thermal balance intermediate hydro-isolated block products for the i-th period of time has the form

$$Q_{IB} = c_{IB} \cdot m_{IB} \cdot (t_{IB\,E} - t_{IB\,B}) = Q_{IBA} + Q_{EXOI} \pm Q_{RWI} \pm Q_{RI} \pm Q_{RBI}, \quad (3.31)$$

where Q_{IB} – the loss of heat for heating the intermediate block products for the i-th period of time J;

c_{IB} – average specific mass heat capacity materials intermediate hydro-isolated block products, J/(kg·ºC);

m_{IB} – mass components of intermediate hydro-isolated block products, kg;

t_{IBB}, t_{IBE} – average temperature components intermediate block products at the beginning and end of the i-th period of time, ºC;

Q_{IBA} – the amount of heat that perceives intermediate block products from heated air for the i-th period of time, J;

Q_{EXOI} – heat-receipts during for the i-th period of time due to the presence of exothermic reaction of hydration of cement in intermediate block products, J; this value is determined from experimental data;

Q_{RWI} – the amount of heat J, transmitted for the i-th period of time due to the presence of resulting radiation from vertical surfaces of the intermediate block of products to the respective wall surfaces camera (or backwards);

Q_{RI} – the amount of heat J, transmitted for the i-th period of time due to the presence of the resulting radiation inverse up horizontal surface intermediate block of products to the inverse down surface located above of the block of products (or backwards);

Q_{RBI} – the amount of heat J, transmitted for the i-th period of time due to the presence of resulting radiation from the inverse down horizontal surface intermediate block of products to the inverse up horizontal surface located below block of products (or backwards).

The direction of these flows resulting radiation depends on the ratio between the temperatures corresponding surfaces.

The amount of heat Q_{IBA}, which perceives an intermediate block of products from hot air for the i-th period of time J, is equal to

$$Q_{IBA} = c_{IB} \cdot m_{IB} \cdot (t_{AIR} - t_{IBB}) \cdot \beta_I, \quad (3.32)$$

t_{AIR} – average for the i-th period of time temperature in the chamber, ºC;

β_I – coefficient of, which takes into account the proportion of heat perceives an intermediate block products for the i-th period of time (relative to the amount of heat needed for the heating block to a temperature t_{AIR}).

Thermal balance lower hydro-insulated block of products for the i-th period of time has the form

$Q_{BL} = c_{BL} \cdot m_{BL} \cdot (t_{BLE} - t_{BLB}) = Q_{ABL} + Q_{EXOL} \pm Q_{RWL} \pm Q_{RL} \pm Q_{RL}$, (3.33)

where Q_{BL} – the loss of heat for heating the lower block products for the i-th period of time, J;

c_{BL} – average specific mass heat capacity materials lower hydro-isolated block of products, J/(kg·°C);

m_{BL} – mass components lower of hydro-isolated block of products, kg;

t_{BLB}, t_{BLE} – the average temperature of the components of the lower block products at the beginning and end of the i-th period of time, ºC;

Q_{ABL} – the amount of heat that perceives lower block of products from heated air for the i-th period of time, J;

Q_{EXOL} – heat-receipts during the i-th period of time due to the presence of exothermic reaction of hydration of cement in the bottom block of products, J;

Q_{RWL} – the amount of heat J, transmitted for the i-th period of time due to the presence of resulting radiation from vertical surfaces of the lower block of products to the respective wall surfaces of the camera (or backwards);

Q_{RL} – the amount of heat J, transmitted for the i-th period of time due to the presence of resulting radiation from the inverse up horizontal surface of the lower block of products to the inverse down surface located above block of products (or backwards);

Q_{RL} – the amount of heat J, transmitted for the i-th period of time due to the presence of resulting radiation from the inverse down horizontal surface to the lower block products to the surface floor (or backwards).

The direction of these flows resulting radiation depends on the ratio between the temperatures corresponding surfaces.

The amount of heat Q_{ABL}, which perceives lower block products from hot air for the i-th period of time J, is equal to

$$Q_{ABL} = c_{BL} \cdot m_{BL} \cdot (t_{AIR} - t_{BLB}) \cdot \beta_L, \quad (3.34)$$

t_{AIR} – average for the i-th period of time temperature in the chamber, ºC;

β_L – coefficient of, which takes into account the proportion of heat perceives lower block of products for the i-th period of time (relative to the amount of heat needed for the heating block of to a temperature t_{AIR}).

The amount of heat transmitted by the i-th period of time due to the presence of the resulting radiation Q_{RWU}, Q_{ROU}, Q_{QRU}, Q_{RWI}, Q_{QRI}, Q_{RBI}, Q_{RWL} ,Q_{RL}, Q_{RL} calculated by the general formula (3.18).

The simplified heat balance chambers for thermal treatment of concrete products (variant 2). If the mass of components of hydro-insulated blocs of products are identical, the difference in temperatures between these blocks of products is negligible. Then when determining the temperature change of concrete products and air temperature in the chamber using the same dependence, and for the camera (variant 1). In calculating these relationships should take into account the difference in the design of the camera.

3.3. Determination of the temperature change of concrete products and changes in air temperature in heating chamber (variant 3)

Scheme third variant waterproofing of concrete products (paving slabs) is shown in Fig. 3.5.

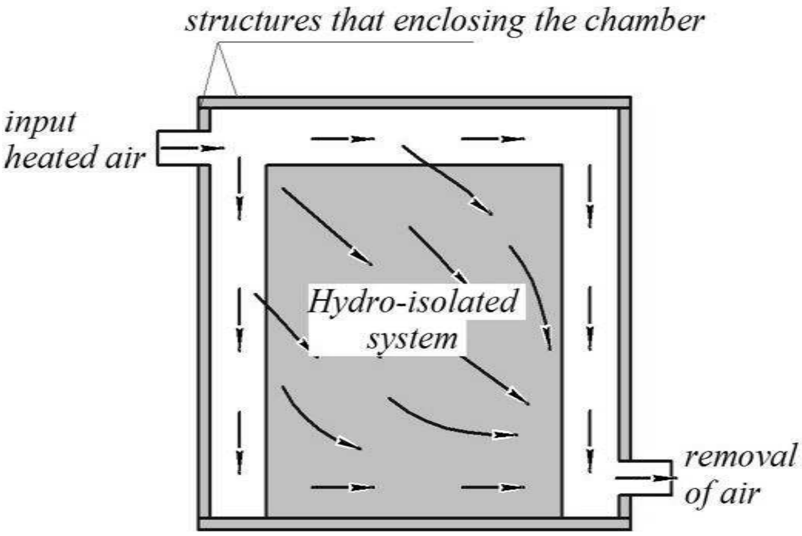


Fig. 3.5. Scheme third variant waterproofing of concrete products

Thermal balance of this camera for thermal treatment of concrete products hot air for the i-th period of time has the form

$$Q_{CE} + Q_{EXO} = Q_{CR} + (Q_S + Q_{LF1} + Q_{S1}) + Q_{CW} + Q_{CO} + \\ + Q_{WA} + Q_{OA} + Q_{FL2} + Q_{S2}, \quad (3.35)$$

where Q_{CE} , Q_{EXO} , Q_{CR} , Q_{CW} , Q_{CO} , Q_{WA} , Q_{OA} – the same the value as in the thermal balance (3.1);

Q_{FL1} – the loss of heat for the i-th period of time for heating the the floor of the chamber, located within the hydro-isolated system, J;

Q_{S1} – the loss of heat for the i-th period of time for heating corresponding the soil J. (provided these losses in the i-th period of time);

Q_{FL2} – the loss of heat for the i-th period of time for heating the the floor of the chamber, located outside of hydro-isolated system, J;

Q_{S2} – the loss of heat for the i-th period of time for heating corresponding the soil, J (provided these losses in the i-th period of time).

Thermal balance chamber (3.35) is made, provided that there are no air vents in the chamber. If there are air vents in the chamber, the heat balance of the camera has the form

$$Q_{CE} + Q_{EXO} = Q_{CR} + (Q_S + Q_{LF1} + Q_{S1}) + Q_{CW} + Q_{CO} + + Q_{WA} + Q_{OA} + Q_{FL2} + Q_{S2} + Q_{AP}, \quad (3.36)$$

where Q_{AP} – heat loss for the i-th period of time and for heating the air pipes camera, J.

Chapter 4. Determination of heat transfer coefficients

4.1. Determination of heat transfer coefficients in convection movement of air in the collector solar energy and in chamber for thermal treatment of concrete products

Calculation:

– heat transfer coefficient α_C from the air, moving in solar energy collector on the respective surfaces of heat transfer,

– heat transfer coefficient α *from the air, heated in the collector solar energy to heat transfer surfaces camera*

in accordance with the methodology reflects the process convective heat transfer in channels with forced air movement. At the presentation of this methods used symbol which is used as primary sources.

The coefficient of heat transfer from the heated air to the surfaces of the channel, W/(m^2·°C), calculated by the formulas given in particular in the sources [27], [28],[29].

Regime of air traffic in the channel established to the value of the Reynolds criterion, which is calculated by the formula

$$Re_{air} = \frac{\omega\, d_{eq}}{\nu_{air}}, \quad (4.1)$$

where ω – speed of air movement in the channel, m/s; d_{eq} – the inner diameter of the channel, m; ν_{air} – kinematic viscosity coefficient of air, m^2/s.

Dependence to calculate an equivalent diameter $d_{ек}$, m, has the form

$$d_{eq} = \frac{4f}{p}, \quad (4.2)$$

where f – cross sectional area of the channel, m^2; p – perimeter cross section of the channel, m.

In turbulent mode air movement heat transfer coefficient $\bar{\alpha}$, W/(m^2·°C), calculated using the formula

$$\overline{Nu_{air}} = \frac{\bar{\alpha}\, d_{eq}}{\lambda_{air}}, \quad (4.3)$$

$$\text{at} \quad \bar{\alpha} = \overline{Nu_{air}} \frac{\lambda_{air}}{d_{eq}}. \quad (4.4)$$

The Nusselt number in turbulent air movement in the channel can be calculated using the formula provided in the sources [27], [28], [30],

$$\overline{Nu_{air}} = 0{,}021\ Re_{air}^{0,8} Pr_{air}^{0,43} \varepsilon_{l}, \tag{4.5}$$

where index p shows that the physical properties of air are determined with the average length of the channel air temperature t_{air}, ºC;

ε_l – coefficient (amendment), taking into account the change in the average heat transfer coefficient along the length of the channel.

In the chamber for thermal treatment of concrete products available change direction of the flow. When calculating the heat transfer coefficient α from the heated air to the surfaces of hydro-isolated system and from the heated air to the inner surfaces of structures enclosing the chamber, you must consider the following provisions contained in the source [28]:

– at the value of criterion of Reynolds $Re_{cr}^{/} < Re < Re_{cr}^{//}$ for calculating criterion of Nusselt $\overline{Nu_{air}}$ can be apply the formula (4.5);

– at $Re > Re_{cr}^{//}$ can be use the formula (4.5), but at use of coefficient ε .

The critical value of criterion of Reynolds of determined by dependencies

$$Re_{cr}^{/} = \frac{16{,}4}{\sqrt{d_{eq}/R}}; \tag{4.6}$$

$$Re_{cr}^{//} = 18500\left(\frac{d_{eq}}{2R}\right)^{0,28}, \tag{4.7}$$

where R – the radius of curvature.

The coefficient ε is equal to

$$\varepsilon = 1 + 1{,}77\ \frac{d_{eq}}{R}. \tag{4.8}$$

In laminar air movement in the channel heat transfer coefficients calculated on dependencies given particular in the sources [27], [28], [30].

4.2. Determination of heat transfer coefficients external surfaces of structures enclosing collector solar energy and chamber for thermal treatment of concrete products

4.2.1. General provisions

The coefficient of heat transfer from the external surfaces of the walls of the chamber to the environment, W/(m^2 ·ºC), calculated by the formula

$$\alpha_{WE} = \alpha_{CW} + \alpha_{RW}; \tag{4.9}$$

The coefficient of of heat transfer from the outer surface of the overlapping camera to the environment, W/(m^2 ·ºC), calculated by the formula

$$\alpha_{OE} = \alpha_{CO} + \alpha_{RO}, \tag{4.10}$$

where α_{CW} – heat transfer coefficient convection of external wall surfaces camera, W/(m^2 ·ºC); α_{RW} – heat transfer coefficient radiation of external wall surfaces camera, W/(m^2 ºC); α_{CO} – heat transfer coefficient convection the outer surface of overlapping cameras, W/(m^2 ·ºC); α_{RO} – heat transfer coefficient radiation the outer surface of overlapping cameras, W/(m^2 ºC).

The coefficients of heat transfer radiation external surfaces of structures enclosing chamber for thermal treatment of concrete products, W/(m^2 ·ºC), calculated in accordance with the recommendations given in particular in the sources [27 [28]:

– for the walls chamber:

$$\alpha_{RW} = \frac{\varepsilon_W \cdot c_0}{t_{WE} - t_E} \cdot \left[\left[\left(\frac{T_{WE}}{100}\right)\right]^4 - \left[\left(\frac{T_E}{100}\right)\right]^4\right], \tag{4.11}$$

– for overlapping the camera:

$$\alpha_{RO} = \frac{\varepsilon_O \cdot c_0}{t_{OE} - t_E} \cdot \left[\left[\left(\frac{T_{OE}}{100}\right)\right]^4 - \left[\left(\frac{T_E}{100}\right)\right]^4\right], \tag{4.12}$$

where T_{WE}, T_{OE} – the average for the selected period of time absolute temperature external surfaces of the walls and overlapping chamber, K; T_E – absolute temperature surfaces shop, K (adopted simplification: the temperature inner surfaces of the room is equal to room air temperature); ε_W, ε_O – degree of blackness materials; c_0 – coefficient of radiation of a blackbody, W/(m^2 ·K^4); ($\Delta t = \Delta T$).

The coefficients of heat transfer from the external surfaces structures of the light-transparent covering solar energy collector to the environment (α_{LH}, α_{LV}) and heat transfer coefficients from the horizontal and vertical outer surfaces of the layer of thermal insulation to the environment (α_{TIH}, α_{TIV}) account for only convective component of the heat transfer process (explanation given in p. 2.2).

4.2.2. Determination of the average for height surface coefficient of heat transfer in natural convection

According to one of the following methods can be calculated:

– heat transfer coefficient convection of external wall surfaces chamber α_{KC};

– heat transfer coefficient convection of external surfaces vertical of structures of light-transparent cover solar energy collector α_{LV} heat transfer coefficient vertically located external surfaces layer of thermal insulation of solar energy collector α_{TIV} (in the absence of wind).

At the presentation of the provisions described methods was used designation heat transfer coefficients convection proposed in the primary sources.

Determination of the average for height surface coefficient of heat transfer in natural convection respectively with the recommendations of sources [27], [28].

Average for height surface coefficient of heat transfer in natural convection $\bar{\alpha}$, W/(m^2·°C), calculated using the formula

$$\overline{Nu_{air}} = \frac{\bar{\alpha}\, h}{\lambda_{air}}, \tag{4.13}$$

$$\text{at} \quad \bar{\alpha} = \overline{Nu_{air}}\, \frac{\lambda_{air}}{h}, \tag{4.14}$$

where h – the height of the heat transfer surface, m; λ_{AIR} – coefficient of thermal conductivity air, W/(m· °C);

Index p shows that the physical properties of air are determined when the air temperature outside the boundary layer t_{air}, °C.

For vertical surfaces in laminar mode movement of air in the boundary layer ($10^3 < Gr_{air}Pr_{air} < 10^9$) according to the recommendations source [28], Nusselt number $\overline{Nu_{air}}$ calculated by the formula

$$\overline{Nu_{air}} = 0{,}75\ (Gr_{air}\, Pr_{air})^{0,25}; \tag{4.15}$$

where Gr – Grashof criterion; Pr – Prandtl number; in the book [27] this formula has the form:

$$\overline{Nu_{air}} = 0{,}76\ (Gr_{air}\, Pr_{air})^{0,25}. \tag{4.15a}$$

in the book [27] proposed to adopt $Pr_{air} \approx 0{,}7$ and simplify the formulas above:

$$\overline{Nu_{air}} = 0{,}695\ Gr_{air}\ ; \tag{4.16}$$

For vertical surfaces in a turbulent mode movement of air in the boundary layer according to the recommendations sources [28], Nusselt number $\overline{Nu_{air}}$ calculated by the formula

$$\overline{Nu_{air}} = 0{,}15\ (Gr_a\, Pr_a)^{1/3}, \tag{4.17}$$

which is used in $Gr_{air}Pr_{air} \geq 6{\cdot}10^{10}$ (developed turbulent motion of air in the boundary layer); in the textbook [28] is noted that when $10^9 < Gr_{air}Pr_{air} < 6{\cdot}10^{10}$ transitional regime is a movement environment in the boundary layer.

In the book [27] The above formula has the form

$$\overline{Nu_{air}} = 0{,}15\ (Gr_{air}\, Pr_{air})^{0,33}, \tag{4.17a}$$

this formula in the book [27] is recommended to apply when $Gr_{air}Pr_{air} > 10^9$; in the book [27] proposed to adopt $Pr_{air} \approx 0{,}7$ and simplify the formulas above:

$$\overline{Nu_{air}} = 0{,}133\ Gr_{air}^{0,33}. \tag{4.18}$$

$Gr_{air}Pr_{air}$ (Rayleigh criterion Ra_{air}) calculated by the formula

$$Gr_{air}\,Pr_{air} = \frac{g\beta_{air}\,\Delta t\,h^3}{\nu_{air}^2}Pr_{air}, \qquad (4.19)$$

where Δt – temperature head, ºC.

Determination of the average for height surface coefficient of heat transfer in natural convection respectively with the recommendations of the handbook [30]. Average for height surface heat transfer coefficient $\bar{\alpha}$, W/(m^2·ºC), in laminar mode movement of air in the boundary layer (Gr_aPr_a = $10^3 \ldots 10^9$) calculated using the formula

$$\overline{Nu_a} = \frac{\bar{\alpha}\,h}{\lambda_a}, \qquad (4.20)$$

$$\text{тоді} \quad \bar{\alpha} = \overline{Nu_a}\,\frac{\lambda_a}{h}, \qquad (4.21)$$

$$\text{where} \quad \overline{Nu_a} = 0{,}8\,(Gr_aPr_a)^{1/4}\varepsilon\,; \qquad (4.22)$$

$$Gr_pPr_p = \frac{g\beta_a\,\Delta t\,h^3}{\nu_a^2}Pr_a\,; \qquad (4.23)$$

$$\varepsilon = \left[1+\left(1+\frac{1}{\sqrt{Pr_a}}\right)^2\right]^{-1/4}; \qquad (4.24)$$

λ_a – coefficient of thermal conductivity air, W/(m· ºC); h – the height of the heat transfer surface, m; g – acceleration of gravity, m/s^2; β_a – temperature coefficient of volume expansion of air, K^{-1}; ν_a – kinematic viscosity coefficient of air, m^2/s;

index p shows that the physical properties of air are determined during the design temperature t_p, ºC, is equal to

$$t_p = 0{,}5\,(t_{air} + t_s); \qquad (4.25)$$

Δt – temperature head, ºC, is equal to

$$\Delta t = t_s - t_{air}, \qquad (4.26)$$

where t_{air} – air temperature outside the thermal boundary layer, ºC; t_s – the temperature of the heat transfer surface, ºC.

The handbook [30] noted that the calculation of heat transfer in vertical surfaces Gr_aPr_a > 10^9 carried out separately for areas with laminar and turbulent boundary layers.

Average for height surface heat transfer coefficient $\bar{\alpha}$, Вт/(м^2·ºC), at Gr_aPr_a > 10^9 calculated by the equation

$$\bar{\alpha} = \bar{\alpha}_l\,\frac{h_{cr}}{h} + \bar{\alpha}_t\left(1 - \frac{h_{cr}}{h}\right); \qquad (4.27)$$

the average heat transfer coefficient on the section of the laminar boundary layer $\bar{\alpha}_l$, W/(m^2·ºC), calculated using the formula

$$\overline{Nu_a} = \frac{\overline{\alpha_l}\, h_{cr}}{\lambda_a}, \tag{4.28}$$

$$\text{at} \quad \overline{\alpha_l} = \overline{Nu_a}\, \frac{\lambda_a}{h_{cr}}, \tag{4.29}$$

where criterion $\overline{Nu_a}$ calculated by the formula (4.22);

the average heat transfer coefficient on the section with turbulent boundary layer $\overline{\alpha_t}$, Вт/(м^2·ºC), calculated using the formula

$$\overline{Nu_a} = \frac{\overline{\alpha_t}\,(h - h_{cr})}{\lambda_a}, \tag{4.30}$$

$$\text{at} \quad \overline{\alpha_t} = \overline{Nu_a}\, \frac{\lambda_a}{h - h_{cr}}, \tag{4.31}$$

$$\text{where} \quad \overline{Nu_a} = 0{,}15\,(Gr_a Pr_a)^{1/3}; \tag{4.32}$$

h – the height of the heat transfer surface, m; $(h - h_{cr})$ – the height sections with turbulent boundary layer, m; h_{cr} – the height sections with laminar boundary layer (the critical the height the heat transfer surface), m, determined from the condition:

$$Gr_a Pr_a = \frac{g\beta_a\, \Delta t\, h_{cr}^3}{\nu_a a_a} = 10^9, \tag{4.33}$$

where a_a – coefficient of thermal diffusivity air, m^2/s.

Heat transfer coefficient convection the outer surface of vertical constructions enclosing equipment for accelerated hardening of concrete products, W/(m^2 ·ºC), in accordance with the recommendations presented in the book [31] , calculated by dependence

$$\alpha_c = 1{,}163 \cdot 2{,}2 \cdot \sqrt[4]{t_w - t_a}, \tag{4.34}$$

where t_w – the temperature of outer surface of the wall, ºC; t_a – air temperature, ºC.

The heat transfer coefficient vertical surface with natural convection, W/(m^2 ·ºC), in accordance with the recommendations presented in the book [32] is calculated by dependence

$$\alpha_c = 1{,}66 \cdot \sqrt[3]{t_w - t_a}. \tag{4.35}$$

4.2.3. Determination of the average heat transfer coefficient of horizontal surface with natural convection

According to one of the following methods can be calculated:

– heat transfer coefficient convection the outer surface of overlapping chamber $\alpha_{КП}$;

– heat transfer coefficient convection the outer surface of of the horizontal design light-transparent cover of solar energy collector $\alpha_{СГ}$, heat transfer coefficient convection of horizontal external surfaces of thermal insulation layer of solar energy collector α_{TT} (in the absence of wind);

– heat transfer coefficient near surface of the floor chamber within the hydro-isolated system (variant 3).

At the presentation of the provisions of these methods were used designation heat transfer coefficients convection proposed in primary sources.

The handbook [30] recommended average heat transfer coefficient of horizontal plate surface is inverted up, receive equal

$$\overline{\alpha_h} = \overline{\alpha_v}, \tag{4.36}$$

plate surface inverted down,

$$\overline{\alpha_h} = 0{,}5\,\overline{\alpha_v}\,, \tag{4.37}$$

where $\overline{\alpha_v}$ – the average heat transfer coefficient of vertical plate, W/(m^2·ºC), calculated by equation (4.21) provided $(GrPr)_a < 10^9$, at the same time defining the size is smaller side plates.

In the book [33] is recommended to calculate average heat transfer coefficient of horizontal plate surface is inverted up (at t_w = const), using formulas:

– at $Ra_a < 10^5$

$$\overline{Nu_a} = 1{,}1\ Ra_p^{1/5}; \tag{4.38}$$

– at $Ra_a > 10^5$

$$\overline{Nu_a} = 0{,}203\ R_p^{1/3}. \tag{4.39}$$

In calculating the Rayleigh criterion of $Ra_a = Gr_aPr_a$ and Nusselt criterion of $\overline{Nu_a}$ the determining size L, m, is equal to

$$L = F/\,P, \tag{4.40}$$

where F – horizontal surface area, m^2; P – perimeter horizontal surface, m.

heat transfer coefficient convection the outer surface of of horizontal constructions enclosing equipment for accelerated hardening of concrete products, W/(m^2 ·ºC), in accordance with the recommendations presented in the book [31], calculated by dependence

$$\alpha_c = 1{,}163 \cdot 1{,}8 \cdot \sqrt[4]{t_w - t_a}\,, \tag{4.41}$$

where t_w – the temperature outer surface of designs, ºC;

t_a – air temperature, ºC.

Coefficient of heat transfer horizontal surface in natural convection, W/(m^2 ·ºC), according to the recommendations given in the book [32] is calculated by dependencies:

– when horizontal heated surface inverted up:

$$\alpha_c = 2{,}26 \cdot \sqrt[3]{t_w - t_a}\,, \tag{4.42}$$

– if a horizontal heated surface inverted down:

$$\alpha_c = 1{,}16 \cdot \sqrt[3]{t_w - t_a}\,, \tag{4.43}$$

where t_w – The temperature outer surface of designs, °C.

4.2.4. Determination of heat transfer coefficients external surfaces of of solar energy collector in the presence of the wind

According to one of the following methods can be calculated:

– heat transfer coefficient convection of external surfaces of vertical and horizontal structures of light-transparent cover of solar energy collector (α_{LV}, α_{LH}) subject to the availability of wind;

– heat transfer coefficient vertical and horizontal outer surfaces of the layer of thermal insulation collector of solar energy (α_{TIV}, α_{TIH}) subject to the availability of wind.

It is proposed to calculate these heat transfer coefficients depending on the direction branch and design of the collector:

– by the method that characterizes forced longitudinal flow around the flat surface air;

– by the method that characterizes forced transverse flow around pipe air (because the collector has the shape of of a parallelepiped, in this case, you must use the equivalent diameter d_{eq}).

At the presentation of the provisions of these methods was used designation heat transfer coefficients convection proposed in primary sources.

Determination of the average heat transfer coefficient when forced to flow around the longitudinal flat surface air. Average along the length of flat surface heat transfer coefficient $\bar{\alpha}$, W/(m^2·°C), calculated using Nusselt criterion of

$$\overline{Nu_a} = \frac{\bar{\alpha}\, l}{\lambda_a}, \tag{4.44}$$

$$\text{at} \quad \bar{\alpha} = \overline{Nu_a}\,\frac{\lambda_a}{l}, \tag{4.45}$$

where l – the length of the flat surface, m; λ_a – coefficient of thermal conductivity air, W/(m·°C); index p shows that the physical properties of air are determined at a temperature the main flow t_a.

In sources of the [27], [30] is recommended to take a critical criterion of Reynolds $Re_a = 5 \cdot 10^5$ the boundary between laminar and turbulent motion mode of the air in the boundary layer.

In laminar mode of motion air boundary layer Nusselt number calculated by the formula given in the sources [27], [30],

$$\overline{Nu_a} = 0{,}66\, Re_a^{0,5} Pr_a^{0,33}\,. \tag{4.46}$$

Reynolds number is equal to

$$Re_a = \frac{\omega_0 l}{\nu_a}, \qquad (4.47)$$

where l – the length of the flat surface, m.

In the turbulent regime movement of air in the boundary layer Nusselt number calculated by the formula given in the sources [27], [30],

$$\overline{Nu_a} = 0{,}037\, Re_a^{0,8} Pr_a^{043}. \qquad (4.48)$$

In the book [27] proposed to use a simplified formula for calculating the criterion of $Nu_п$:

– at laminar movement of air in boundary layer

$$\overline{Nu_a} = 0{,}57\, Re_a^{0,5}; \qquad (4.49)$$

– at turbulent movement of air in boundary layer

$$\overline{Nu_a} = 0{,}032\, Re_a^{0,8}. \qquad (4.50)$$

Determination of the average perimeter channel heat transfer coefficient when forced to flow around the transverse channel air. According to the recommendations of the textbook [28] on the perimeter tubes of the average coefficient of heat transfer $\overline{\alpha}$, W/(m^2·°C), calculated using the formula

$$\overline{Nu_a} = \frac{\overline{\alpha}\, d_{ext}}{\lambda_a}, \qquad (4.51)$$

$$\text{at} \quad \overline{\alpha} = \overline{Nu_a}\, \frac{\lambda_a}{d_{ext}}. \qquad (4.52)$$

For the collector this dependence can be written as

$$\overline{\alpha} = \overline{Nu_a}\, \frac{\lambda_a}{d_{eq}}, \qquad (4.53)$$

where d_{eq} – equivalent diameter, m.

The Nusselt number calculated by the formulas:

а) при $5 < Re_a < 10^3$ $\quad \overline{Nu_a} = 0{,}5\, Re_a^{0,5} Pr_a^{0,38};$ (4.54)

б) при $10^3 < Re_a < 2 \cdot 10^5$ $\quad \overline{Nu_a} = 0{,}25\, Re_a^{0,6} Pr_a^{0,38};$ (4.55)

в) при $Re_a = 3 \cdot 10^5 \ldots 2 \cdot 10^6$ $\quad \overline{Nu_a} = 0{,}023\, Re_a^{0,8} Pr_a^{0,37},$ (4.56)

where λ_a – coefficient of thermal conductivity air, W/(m· °C);

Reynolds number is equal to

$$Re_a = \frac{\omega_0\, d_{ext}}{\nu_a}, \qquad (4.57)$$

Dependence of Nu_a are valid when the angle ψ between the direction of the flow and the axis of the tube (channel) is 90^0. With decreasing angle ψ heat transfer intensity decreases.

Average on the perimeter tube heat transfer coefficient at $\psi < 90^0$ $\overline{\alpha_\psi}$, W/(m^2·°C), calculated by dependence

$$\overline{\alpha_{\psi}} = \overline{\alpha}\,\varepsilon_{\psi}, \qquad (4.58)$$

where $\overline{\alpha}$ – average on the perimeter tube (channel) heat transfer coefficient defined for angle $\psi = 90^0$, W/(m$^2 \cdot$ °C);

ε_{ψ} – an amendment to the angle of attack ψ; according to the recommendations source [28] – provided $\psi = 30^0 \ldots 90^0$,

$$\varepsilon_{\psi} = 1 - 0{,}54\cos^2\psi\,. \qquad (4.59)$$

Chapter 5. Determination of heating intensity constructions enclosing chamber for thermal treatment of concrete products and thermal insulation layer of solar energy collector

5.1. The definition changes over time temperature distribution in structures that enclosing chamber for thermal treatment of concrete products and thermal insulation layer of solar energy collector

During the period thermal treatment of concrete products is not only heating these products, and - heating constructions enclosing camera and of solar energy collector. Suppose that the temperature structures of camera and collector insulation layer changes only in the direction normal to their surfaces.

The mathematical characteristic non-stationary heat conduction process in single layer design with one-dimensional temperature field consists of a differential equation of heat conduction

$$\frac{\partial t}{\partial \tau} = a \cdot \frac{\partial^2 t}{\partial x^2}\,, \qquad (5.1)$$

and the uniqueness of the conditions (geometrical, physical, initial and boundary).

Adopted boundary conditions of the third kind: for wall and overlapping Camera, for indoor surface of the floor chamber and to the outer surface of the insulation of the collector solar energy. For the internal surface of the insulation layer is assumed that the surface temperature is equal to the temperature of the metal layer. The floor of chamber warms and heats the soil. For chamber in question proposed simplification: soil taken as conventional component of a floor.

Solve the differential equation (5.1) proposed FDTD method, which are, in particular, in the book [34]. This source also recommendations on the application of this method for a multilayer structure.

In FDTD equation (5.1) has the form

$$\frac{\Delta \mathrm{t}}{\Delta \tau} = a \cdot \frac{\Delta^2 \mathrm{t}}{\Delta \mathrm{x}^2}\,. \qquad (5.2)$$

The construction (wall, floor or overlapping camera, layer the insulation of the collector solar energy) conditionally divided into layers thick Δx, and time – on intervals $\Delta\tau$.

Temperature and i-th plane design $t_{i,\tau+1}$ at the time $(\tau+\Delta\tau)$ calculated by dependence

$$t_{i,\tau+\Delta\tau} = t_{i,\tau} + a \cdot \frac{\Delta\tau}{\Delta x^2} \cdot (t_{i+1,\tau} + t_{i-1,\tau} - 2t_{i,\tau}), \tag{5.3}$$

where $t_{i-1,\tau}$, $t_{i,\tau}$, $t_{i+1,\tau}$ – temperature respectively (i–1)-th, i-th та (i+1)-th planes design in moment in time τ, ºC;

a – coefficient of thermal diffusivity material of construction, m^2/c.

The coefficient of thermal diffusivity material, m^2/c, calculated by the formula

$$a = \frac{\lambda}{c\rho}, \tag{5.4}$$

where λ – coefficient of thermal conductivity material, W/(m·ºC); c – specific mass heat capacity material, J/(kg·ºC); ρ – density material, kg/m^3.

If you select a time interval $\Delta\tau$ and Δx so that the condition

$$\frac{2a\,\Delta\tau}{\Delta x^2} = 1, \tag{5.5}$$

then the temperature in the i-th plane design in moment in time $(\tau+1)$ can be determined by the equation

$$t_{i,\tau+\Delta\tau} = \frac{t_{i+1,\tau} + t_{i-1,\tau}}{2}. \tag{5.6}$$

The maximum period of time $\Delta\tau_{max}$ for use equation (5.6) is equal to

$$\Delta\tau_{max} = \frac{\Delta x^2}{2a}. \tag{5.7}$$

The temperature of the inner surfaces of the walls of chamber, the inner surface of overlapping chamber and floor chamber at the time $(\tau+\Delta\tau)$ calculated by the *total* dependence

$$t_{1,\tau+\Delta\tau} = \frac{\alpha_1 \cdot t_a + \frac{\lambda}{\Delta x} \cdot t_{2,\tau}}{\alpha_1 + \frac{\lambda}{\Delta x}}, \tag{5.8}$$

where $t_{1,\tau+\Delta\tau}$ – the temperature of the first plane design in moment in time $(\tau+\Delta\tau)$, ºC; t_a – the air temperature in the chamber, ºC; $t_{2,\tau}$ – temperature 2nd plane design in moment in time τ, ºC; α_1 – heat transfer coefficient of internal surfaces of structures chamber, W/(m^2 ·ºC).

In determining the heat transfer coefficient α_1 internal surfaces of structures chamber should be considered:

– the transfer of heat from the heated air to the inner surface of these structures by forced convection;

– the presence of radiation heat transfer between these surfaces and hydro-isolated system (hydro-isolated blocks of products).

The temperature of last ((n+1)-th) plane design (the outer surface of the walls and overlapping of the camera and the outer surface of layer the insulation of solar energy collector) at the time ($\tau+\Delta\tau$) calculated by the *total* dependence

$$t_{n+1,\,\tau+\Delta\tau} = \frac{\alpha_2 \cdot t_{AT} + \dfrac{\lambda}{\Delta x} \cdot t_{n,\,\tau}}{\alpha_2 + \dfrac{\lambda}{\Delta x}}, \tag{5.9}$$

where n – the number of layers that are conventionally divided design; t_{AT} – ambient temperature, ºC; $t_{n,\tau}$ – temperature n-th plane design in moment in time τ, ºC; α_2 – coefficient of of heat transfer from the outer surface of the design for the environment, W/(m^2 ·ºC).

5.2. Determining the amount of heat expended on heating designs that enclosing chamber for thermal treatment of concrete products and thermal insulation layer of solar energy collector

The loss of heat for heating a single-layer structure (wall, floor or overlapping camera, the insulation layer of the collector solar energy) for a certain period of time calculated by the formula

$$Q = \sum_{i=1}^{i=n} c \cdot m \cdot (t_{E_i} - t_{I_i}), \tag{5.10}$$

where n – the number of layers that are conventionally divided structure to determine the temperature distribution in it; c – specific mass heat capacity material structures, J/(kg ·ºC); m – mass i-th conventional layer design, kg (mass conventional of layers of design similar); t_{I_i} – initial temperature i-th conventional layer design, ºC; t_{E_i} – temperature i-th conventional layer construction at the end of the selected period of time, ºC.

Temperature i-th conventional layer construction at the end of the selected period of time is calculated by the equation

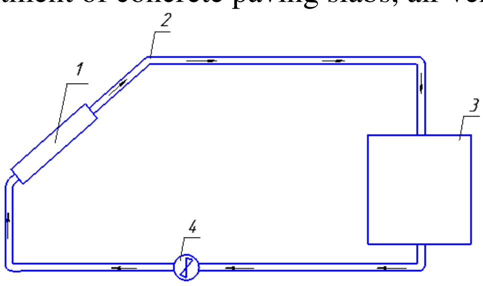
$$t_{C_i} = 0{,}5 \cdot (t_i + t_{i+1}), \tag{5.11}$$

where t_i – the temperature at the boundary climbing (i–1)-th and the i-th conventional layers of construction at the end of the selected period of time, ºC; t_{i+1} – temperature on the boundary climbing i-th and (i+1)-th conventional layers of construction at the end of the selected period of time, ºC.

In determining the temperature distribution in the multilayer structure (a wall, a overlapping or a floor chamber) for conventional layers each layer divided structure. The cost of heat for heating multilayer structures are calculated taking into account the availability of conditional layers.

Chapter 6. Calculation of intensity temperature change of concrete products and air in equipment

We consider the installation, which includes: collector of solar energy, chamber for heat treatment of concrete paving slabs, air vents, fan (Fig. 6.1).



1 - collectors of solar energy; 2 - air vents; 3 - chamber for thermal treatment hydro-insulated concrete products; 4 - fan

Fig. 6.1. Scheme heat supply chamber for heat treatment of concrete paving slabs using heated in collector of solar energy air

Determination of change air temperature in the collector of solar energy. Table. 6.1 shows the results of calculation of air temperature changes in the collector of solar energy in the first 15 minutes of its use in conditions city of Poltava in June (at different values of air consumption L, m^3/h). Receipt of to solar radiation of the collector surfaces defined according to [35]. Accepted: wind speed near of the collector equal to zero; the average for the first period of time the air temperature at the inlet to the collector t_{AI} = 20ºC. Determined by iterations (successive approximation) average for the first period of time the air temperature at the outlet of the collector t_{AO}.

The calculation was made under a simplified procedure. In the detailed method of determining intensity changes in air temperature in the collector solar energy necessary to make thermal balance sheets: light-transparent cover; metal plate and of thermal insulation layer; of the collector solar energy. In the simplified method the average for the i-th period of time temperature light transparent coating is not determined by means of appropriate thermal balance, and taken equal:

$$t_A = 0{,}5\cdot(t_{AIR} + t_{AT}), \qquad (6.1)$$

where t_{AIR} – average for the i-th period of time air temperature in the collector, ºC; t_{AT} – ambient temperature, ºC.

Table 6.1. The results of calculating air temperature change in the collector of solar energy in the first 15 minutes

L, m^3/h	The length and width of the metal plate, m	T_{AI}, ºC	T_{AO}, ºC	$T_{AO} - t_{AI}$, ºC
90	1×1	20	29,5	9,5
135	1×1	20	27,5	7,5
180	1×1	20	26,3	6,3
240	1×1	20	25,2	5,2
355	1×1	20	23,9	3,9
950	1×1	20	21,8	1,8
1440	1×1	20	21,2	1,2
1880	1×1	20	21,0	1,0

Table 6.2 shows the results of calculation temperature change concrete paving slabs and air in the chamber for the first 15 minutes of work equipment for different air flow L, m^3/h. The total mass of concrete in the chamber is equal to 279.5 kg. Accepted: initial temperature of the components of hydro-isolated system $t_{ГП}$ (initial temperature of concrete products t_{BI}) is equal to 20ºC; the average for the first period of time the temperature of air entering the camera, is equal to t_A = 25 ºC. Determined by iterations (successive approximation): the average for the first period of time air temperature, remove from the chamber, t_{AR}; average temperature hydro-isolated system (which includes concrete products) at the end of the first period of time t_{HIE} = t_{CE} ($t_{БК}$ – concrete products temperature end of the first period of time).

Table 6.2. The results of calculation temperature change concrete paving slabs and air in the chamber for the first 15 minutes of work equipment

L, m^3/h	T_A, ºC	T_{AE}, ºC	T_A, ºC	T_{CB}, ºC	T_{CE}, ºC	$T_{CE} - t_{CB}$, ºC
90	25	20,8	22,90	20	20,56	0,6
135	25	21,2	23,10	20	20,67	0,7
180	25	21,5	23,25	20	20,78	0,8
240	25	21,7	23,35	20	20,91	0,9
355	25	22,0	23,50	20	21,13	1,1
950	25	22,9	23,95	20	21,94	1,9
1440	25	23,2	24,10	20	22,35	2,4
1880	25	23,5	24,25	20	22,65	2,7

Table 6.3 shows the results of calculation of air temperature change and concrete paving slabs to install for the first 15 minutes of work. It is simplification: the change of temperature in the heat-insulated air vents and a fan neglected.

Table 6.3. The results of calculating changes in air temperature and concrete paving slabs for the first 15 minutes of work equipment

L, m^3/h	$t_{ПВ}$ = = t_A, °C	$t_{ПН}$ = = $t_{ПК}$, °C	$t_{ПВ}$ – –$t_{ПН}$, °C	t_A, °C	t_{CB}, °C	t_{CE}, °C	t_{CE} – – t_{CB}, °C
90	31,0	22,2	8,8	26,6	20	20,89	0,9
950	24,9	23,0	1,9	23,95	20	21,94	1,9
1880	24,0	22,9	1,1	23,45	20	22,21	2,2

By using heat balances of the collector solar energy and chambers for thermal treatment of concrete products can determine the impact of certain design solutions chamber on the intensity of heating concrete products.

Conclusions

Developed thermal balance flat of solar energy of the collector designed for the heating air. The heated air is used for heat treatment of concrete products.

Developed thermal balance chambers for thermal treatment of concrete products heated air.

Analysis of heat transfer processes occurring in the collector of solar energy and in the chambers for thermal treatment for concrete products, allows you to:

– adopt optimal constructive solutions of this equipment and determine the optimum amount of air circulating in it;

– to perform approximate forecasting intensity temperature change of concrete products in the equipment.

In calculation model of solar energy of the collector horizontal coverage and sides of of the collector is made of transparent material. In the future, we must analyze the processes of heat transfer in a flat the collector provided the sides of of the collector have a different design solution.

In the study of heat transfer features of in chambers for thermal treatment of concrete products hot air taken that in the hydro-isolated system thickness of the air layer the minimum, so you can ignore: the presence of mass transfer processes between open product surface and air layers; the influence of the air layer on the processes of heat transfer in hydro-isolated system. In further studies should analyze the impact of the presence of the air layer in hydro-isolated system in these processes.

In a further need to make the thermal balance of the equipment, which includes: collector of solar energy, air vents (Electroheaters) camera for thermal treatment of concrete products, air vents, fan. The purpose of drafting this heat balance - definition of thermal power air vents to provide required temperature regime solidification of concrete products.

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